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"English Electric" electro-magnetic slip coupling.

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The "English Electric" Magnetic Slip Coupling for Boiler House Fan Drives.

General Requirements of Fan Drives.

The efficient operation of boiler plant requires rapid and accurate response of the boiler evaporation to meet changes in the steam demand, which might be subject to wide and frequent variations. Since the modern boiler has a comparatively small water capacity a change in steam demand requires an immediate change in the volume of air handled by the induced draught and forced draught fans to avoid variations in steam pressure. Again close control of air quantity is necessary for efficient fuel combustion, and if made available has a beneficial effect on the coal bill. It is essential therefore that the fans and their drives should be so designed that easy and accurate control of the volume of air can be obtained. The drives should also be efficient and should require the minimum amount of maintenance.

The simplest and most efficient drive consists of a single-speed squirrel-cage motor coupled directly to the fan, but if this is used the control of the air quantity must be carried out mechanically by means of dampers or inlet vanes. Dampers are heavy to operate and are inefficient ; vanes are more efficient but are mechanically complicated and consequently require a considerable amount of maintenance. In addition, with this type of drive the fan runs at a constant speed which must be sufficiently high to enable it to deliver the maximum quantity of air required under overload conditions with a dirty boiler, and prolonged operation at such speeds results in erosion and clogging of the induced draught fan blades.

An improvement is obtained by using a vane-controlled fan driven by a two-speed motor. By this means improved efficiency can be obtained at low volumes, and the fan operates at reduced speed for part of the time, thus reducing the amount of erosion on the blades. The control, however, becomes more complicated since the vanes must be moved from one extreme position to the other each time the speed is changed and the ranges of output at the two speeds must overlap appreciably, particularly if the volume of air normally required is approximately that obtained at the change speed point. The transition from one speed to the other with simultaneous adjustment of the vanes may result in an undesirable surge in the air system, and may take some considerable time.

The disadvantages of this system are overcome if the control of air quantity is carried out entirely by variation of the fan speed. The drive may then consist of either a variable-speed motor, or a constant-speed motor with some form of variable speed device. However, care must be taken to avoid undue complication of the equipment with resultant high maintenance costs, while the speed must be capable of adjustment in small steps to give as accurate control as possible.

A simple form of drive consists of a slipring motor with rotor resistance control. This, however, has two disadvantages ; unless a liquid speed regulator is used the control gear is complicated for adjustment in fine steps, and either the control gear and resistance must be mounted close to the motor or, if they are separated, heavy cables must be run between the motor, resistance and control board.

This type of control is more efficient than might at first be thought as, due to the torque/speed characteristic of the fan, the maximum extra heat which has to be dissipated in the rotor circuit due to the speed control is about 15 per cent. of the power required by the fan at top speed, and occurs at two-thirds of top speed. At lower speeds the losses decrease until at, say,

one-third speed the rotor loss is 7.4 per cent. of the full-speed fan power.

A drive which has been extensively used consists of a single-speed squirrel-cage motor driving through a scoop-controlled hydraulic coupling, and this combination has only a slightly lower efficiency at the top fan speed and almost the same loss as the slipring motor at lower speeds. This system lends itself to fine adjustment of the fan speed, and since the motor can be started up light it permits the use of a low resistance, low starting torque, squirrel-cage motor, but it suffers from the complication of oil glands and oil coolers, difficulties in remote or automatic control, tendency to instability at low speeds, its characteristics are more affected by ambient temperature changes, and has a slow response to any alteration in the setting. The English Electric Company have, therefore, developed an electro-magnetic slip coupling which, combined with a squirrel-cage motor, gives a variable-speed drive with similar operating losses, but eliminates the disadvantages.

Construction of the Magnetic Coupling.

The construction of the principal parts of the "English Electric" slip coupling is shown in *frontispiece*. The inner field or primary member, shown on the left of the illustration, has an annular field coil co-axial with the driving shaft, with pole rings on either end of it, the pole rings being extended in the form of fingers over the outside of the coil, and the fingers from the two rings being interleaved. The coil is excited by direct current via two sliprings so that one pole ring has a North polarity and the other a South polarity. As the fingers are interleaved this gives the effect of a large number of alternate North and South poles round the periphery.

The primary member runs within the outer or secondary member, shown on the right of the illustration, a small air-gap being left between the two. This secondary is a solid steel ring, the inner surface of which has axial slots which do not extend the full length of the ring, so that the effect is somewhat similar to a squirrel-cage with bars and end rings; there are not, however, any windings in the secondary. The outer surface has circumferential ribs to assist cooling, and a deep groove in the centre cuts into the slots on the inner surface so that air can pass right through the ring in a radial direction.

One of these members is coupled to the motor shaft and the other to the fan shaft, the coupling being reversible in the sense that it does not matter which is the input side and which the output side. The arrangement usually adopted for fan drives is shown in Fig. 1. A sleeve mounted directly on the motor shaft carries the secondary ring by means of a disc in which holes are cut to allow ingress for cooling air. The primary member is carried on ball and roller bearings which run on the outside of this sleeve, and an extension of the hub of the primary carries the sliprings and is also suitable for bolting on a flange-mounted flexible coupling.

With this type of construction the magnetic coupling is a complete unit assembled at the works so that concentricity of the primary and secondary members is ensured and the air gap will not be affected by any small misalignment between the motor and fan shafts. The motor and magnetic coupling in effect form a unit which can be flexibly coupled to the fan just as easily as a motor without a magnetic coupling. Fig. 2 shows a coupling mounted in this manner between two shaft ends.

It will be noted that with this arrangement the secondary ring in which the slip losses have to be dissipated is always running at the same speed as the driving motor, regardless of the fan speed. This materially assists cooling.

As the axial length of the "English Electric" coupling is short, the complete coupling can, in the sizes normally required for boiler house fan drives, be quite safely overhung on the driving

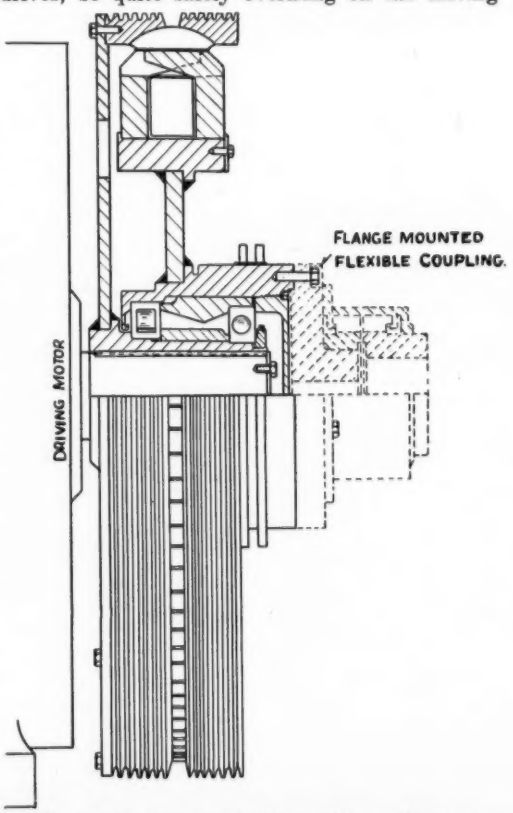


Fig. 1.—Electro-magnetic slip coupling overhung on driving motor shaft.

Fig. 2.—Magnetic coupling mounted between two machines.

motor shaft end without unduly stressing the shaft or bearings. The load on the shaft is, in fact, for the majority of cases, considerably less than that due to the belt pull for a belt drive of the same output.

Operation.

If one member of the coupling is running at motor speed and the field coil is excited, the other member being stationary, eddy currents are set up in the secondary due to the frequent reversals of magnetic flux at each point on the inner surface of the secondary. These eddy currents depend on the intensity of the flux and on the difference in speed between the two members, i.e., the slip speed or slip frequency.

The interaction between the eddy currents and the magnetic flux produces a torque between the two members, this torque being a function of the excitation and the slip speed.

The torque so produced causes the output member to accelerate, thus reducing the slip speed and consequently reducing the torque. The output member continues to accelerate until the combination of flux and slip speed results in a torque which just balances the torque required to drive the fan or other load, after which the speed remains steady until either the load torque or the excitation s altered.

If the coupling is running under steady load conditions and the excitation is reduced a larger difference in speed, or slip, will be required to enable the coupling to transmit the required torque. Since the driving motor speed is approximately constant the speed of the output member must decrease and the slip increase to such a value that the transmitted torque and the load torque again balance. Speed control of the output member can, therefore, be obtained by a simple control of the coupling excitation.

The torque of the motor is transmitted to the driven side unaltered, as a magnetic coupling is obviously not a torque convertor.

Characteristics.

Typical torque/speed curves for an " English Electric " magnetic coupling are shown in Fig. 3. The general shape of the curve is similar to that of a high-resistance squirrel-cage motor, the maximum torque being available when starting from rest. Ample torque is therefore available for accelerating the high inertia of a fan when starting.

The slip at normal load is, however, comparable with that of a low-resistance squirrel-cage motor, but, of course, is additional to the slip of the driving motor.

The excitation requirements of a magnetic coupling on a fan load, are shown in Fig. 4.

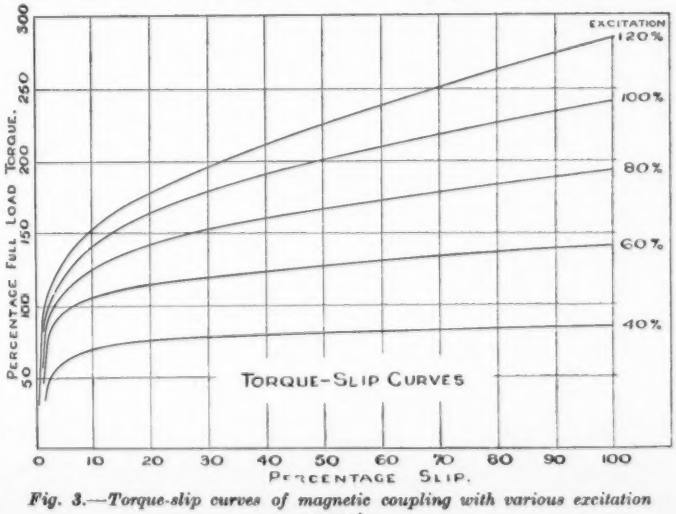


Fig. 3.—Torque-slip curves of magnetic coupling with various excitation currents.

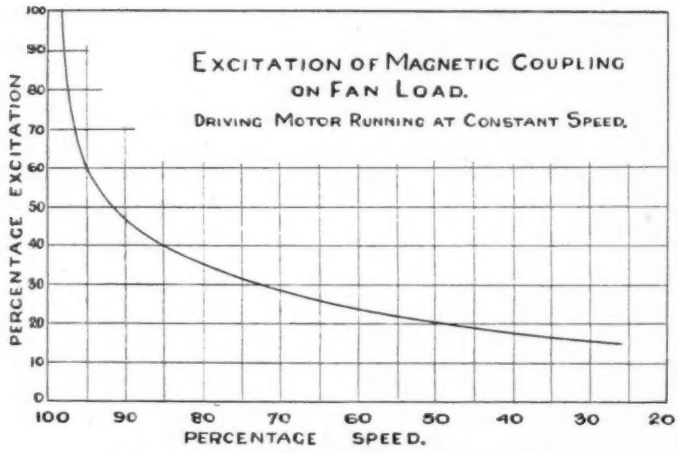


Fig. 4.—Excitation of magnetic coupling on fan load.

A great advantage of this type of magnetic coupling is the quick response of torque to changes in excitation. Oscillograph tests show that with the input member of the coupling running at normal speed and the output member stationary, if normal excitation current be switched on the torque rises to normal value in under one second. Under normal operating conditions such a wide change in excitation is seldom required, so that it can be said that any ordinary adjustment is almost instantaneous. The response of the fan speed is, of course, slower owing to the inertia of the fan runner.

Owing to the continual reversal of the flux there is no permanent magnetisation effect in the secondary member. Thus excitation, torque and speed, retain a fixed relationship only slightly affected by temperature variations in the secondary member. If the speed is altered by a change in excitation and the excitation is then restored to its initial value the speed will also return to its initial value regardless of the magnitude or direction of the change. This is a very valuable feature for automatic control.

Application to Boiler House Fan Drives.

The " English Electric " magnetic coupling is very suitable for the driving of boiler house fans. Its short overall length, particularly when the coupling is overhung on the motor shaft, gives a very compact drive and very little space is required for auxiliary control equipment. The control is carried out by means of a rheostat which need not be located near the coupling, but may be very conveniently mounted on the control panel. Since the excitation requirements are very low—approximately 840 watts, or say, 4 amps. at 220 volts for a 180 H.P., 710 R.P.M. coupling—the rheostat can be small and light and consequently it is easily controlled by automatic equipment, and the exciting current can be supplied from a metal rectifier.

Magnetic couplings are suitable for combined or automatic control, as the couplings for all the drives on one boiler can be operated together by a master rheostat, with trimming rheostats for individual couplings. Where the fuel feed drive is by means of a variable-speed motor, it is easy to link together the regulation of fuel and air supply.

The ribbed secondary system of the magnetic coupling is so designed that it will dissipate naturally the heat loss incurred by any speed variation on a fan drive which rises to its maximum at two-thirds speed but is progressively lower for both higher and lower speeds, just as in a resistance-controlled slipring motor drive.

As conversion equipment with its attendant losses is not required for the motor power, and the exciting power is small, this type of drive compares favourably with other types of variable-speed fan drive and in addition takes up less space, is simple, and only requires low-resistance squirrel-cage motors since starting is carried out under no-load conditions.

Magnetic slip couplings are inherently suitable for any range of speed on fan drives and are quite stable even at crawling speeds. If installed for individual manual control they can very easily be converted to combined or automatic operation at a later date.

Summary of Advantages.

The advantages of the system of boiler house fan drives embodying " English Electric " magnetic slip couplings may be summarised as follows :—

(1) Standard A.C. squirrel-cage motors may be used with the most robust type of windings and starting gear, since they start on no load.

(2) The equipment is mechanically robust and reliable and requires less maintenance than other systems.

(3) Both the motor and coupling are suitable for either a small or a wide speed variation, the only difference being the grading of the small field regulator.

(4) The space taken by the equipment is reduced to a minimum and is less than even a squirrel-cage motor with a hydraulic coupling.

(5) The power for the fan drives may be obtained simply from unit or station transformers or from the house alternator. No D.C. conversion apparatus is required except for the small control and field circuits.

(6) The efficiency compares favourably with

that of a variable speed D.C. motor drive or a constant speed drive with vane or damper control. The loss is greater than with an A.C. commutator motor having induction regulator control, at certain speeds, but it will show to better advantage at higher and lower speeds. It has compensating advantages in simplicity, cost, space required, elimination of the induction regulator with its large volume of oil and fire risk, and better overall power factor.

(7) The system is suitable for individual, combined, or automatic control and the control torque required to operate the field rheostats is less than with any other system.

(8) Rapid, smooth, and consistent response is given to any change in the control.

(9) Commutators are eliminated.

(10) Special cooling arrangements are not required as the magnetic coupling can look after its own heat dissipation.

Torsional Vibration in Internal Combustion Engines.

By L. BERMAN, M.A., A.M.I.Mech.E., Internal Combustion Engine Department, Rugby.

General.

Many types of machinery can be adversely affected by vibration of one kind or another and this is an important aspect in the design of internal combustion engines. The particular type of vibration most likely to cause trouble in this case is a torsional one, in which the various rotating masses (cranks, flywheel, generator, etc.) oscillate in the plane of rotation, i.e., the crankshaft twists first in one direction, then in the other, this motion being superimposed on the steady rotation of the system.

In theory, a system composed of n masses with n-1 connecting shafts will have n-1 modes of vibration, corresponding to 1, 2, 3 . . . n-1 nodes in the shafting, as shown in Fig. 1. In practice, however, only the first few modes need be considered, since the frequency of the others will be very high and they will not readily be excited. The chief difficulty arises from the fact that there are exciting impulses of numerous frequencies, causing resonance at many different running speeds. It is well known that the torque produced by any reciprocating engine is far from uniform and this is particularly marked in the case of internal combustion engines ; the torque curve from any cylinder can therefore be split up into a Fourier series of harmonics of progressively higher frequency, each of which will be in resonance with each vibration frequency at a particular speed. For instance, suppose the fundamental natural frequency of the system to be 4,000 cycles/min. Then the 1st, 2nd, 3rd . . . harmonics of the torque curve will tend to cause vibration when the engine is running at 4,000, 2,000, 1,333 . . . R.P.M. respectively, these being called the critical speeds of the 1st, 2nd, 3rd . . . orders. Four-stroke engines will also have critical speeds of the 1/2, 1½, 2½ . . . orders, since the complete torque cycle for one cylinder occupies two revolutions.

In the earlier stages of the development of the internal combustion engine, the number of cylinders was usually small and the running speed was low, so that the critical speeds occurring in the running range were of a relatively high order number, the corresponding torque harmonics being small ; if any vibration did occur, it was therefore small in amplitude. The trend of design, however, led to the use of greater numbers of cylinders, which lowered the vibration frequencies, and at the same time running speeds rose ; consequently critical speeds of smaller order numbers and greater severity were encountered. The serious difficulties which arose led to a thorough examination of the problem, and methods have gradually been evolved for calculating the characteristics of a given system and thereby ensuring satisfactory running. The design of a crankshaft is now based largely on

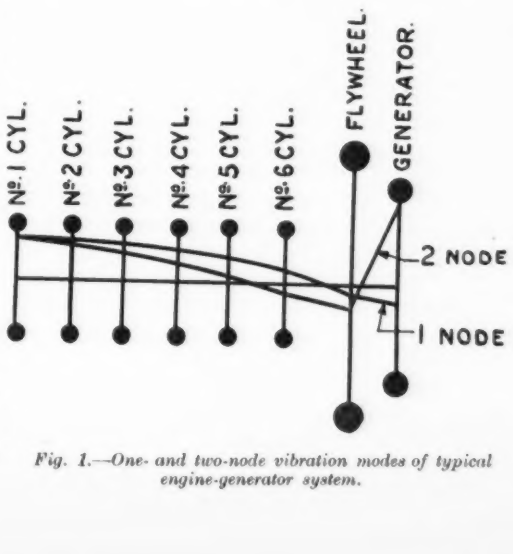


Fig. 1.—One- and two-node vibration modes of typical engine-generator system.

such considerations and it is found that if it is stiff enough to keep the vibration frequency sufficiently high, it will usually be more than strong enough to withstand the normal loads.

Calculation of Natural Frequency.

The polar moments of inertia of the various rotating masses are readily determined, and an allowance for the reciprocating masses can be made. As for the torsional stiffness of the shafting between the masses, any plain cylindrical portions are easily dealt with, but it will be appreciated that the complex structure of a crankshaft is scarcely suitable for accurate mathematical treatment. Empirical formulae have been devised, however, as a result of practical observation, and calculated frequencies can be expected to be correct within fairly close limits. A widely-used formula, given by Carter, states that the torsional stiffness of the crankshaft element shown in Fig. 2 is the same as that of an "equivalent length" l of straight shafting of journal size where

$$l=(2a+0.8h)+\frac{3b}{4}\cdot\frac{D_1^4-d_1^4}{D_2^4-d_2^4}+\frac{3r}{2}\cdot\frac{D_1^4-d_1^4}{hw^3}$$

The frequencies can be determined in various ways when the system has been analysed as described above; the most commonly used method also gives the relative oscillation amplitudes of the various masses, these being required for further calculations. This method depends on the construction of a table, as shown below, in which the amounts of torque and twist in each portion of shaft are progressively determined for an assumed amplitude at one end of the system and an assumed frequency. The last figure in the table is the fluctuating torque which would have to be applied externally to the last mass to maintain the prescribed vibration. By trial, a frequency can be found for which this torque is zero and this is accordingly a frequency of free vibration of the system. Any required number of

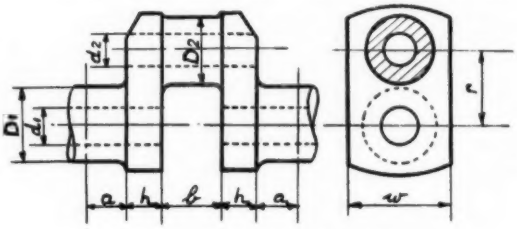


Fig. 2.—Crankshaft element.

vibration modes can be investigated in this manner.

In the case of engine-driven generator sets, it is laid down in B.S.S. 649 that the generator maker shall supply to the engine maker the information necessary for torsional vibration calculations, which are the latter's responsibility. This procedure is followed in the case of all diesel generating sets supplied by the Company; it is often necessary to make a slight alteration to the original generator shaft design to achieve the best running conditions.

Calculation of Amplitudes.

At any particular critical speed, a state of equilibrium obtains between the following:—

(1) Energy input, proportional to the magnitude of the torque harmonic in question and to the vector sum of the vibration amplitudes at the various engine cylinders. This vector sum is different for different orders and consequently only a few of the numerous critical speeds cause vibration of appreciable magnitude, the remainder being insignificant. Typical vector sums for a six-cylinder four-stroke engine are shown in Fig. 3. It will be seen that the largest sum is given by orders 3, 6, 9, 12 . . . these being known as "major orders."

(2) Energy absorption in system. It was thought at first that the principal item was the hysteresis loss in the shaft material caused by the cyclic variation of stress, but later

Frequency Calculation Table.

Assumed Frequency 70.1 cycles/sec.

$$(2\pi \times \text{freq.})^2 = .194 \times 10^6.$$

Mass.	M.I., lb. in. sec.^{2}	M.I. × $.194\times10^6$.	Amplitude, radian.	Accelerating torque, lb. ins.	Total acc. torque, lb. ins.	Shaft stiffness, lb. ins./rad.	Twist in shaft, radian.
No. 1 ...	33.57	6.51×10^6	1.000	6.51×10^6	6.51×10^6	100.3×10^6	.065
No. 2 ...	33.57	6.51×10^6	.935	6.09×10^6	12.60×10^6	100.3×10^6	.126
No. 3 ...	33.57	6.51×10^6	.809	5.27×10^6	17.87×10^6	100.3×10^6	.178
No. 4 ...	33.57	6.51×10^6	.631	4.11×10^6	21.98×10^6	100.3×10^6	.219
No. 5 ...	33.57	6.51×10^6	.412	2.68×10^6	24.66×10^6	100.3×10^6	.246
No. 6 ...	33.57	6.51×10^6	.166	1.08×10^6	25.74×10^6	90.8×10^6	.284
Flywheel...	1,148	223×10^6	—.118	-26.30×10^6	-0.56×10^6		

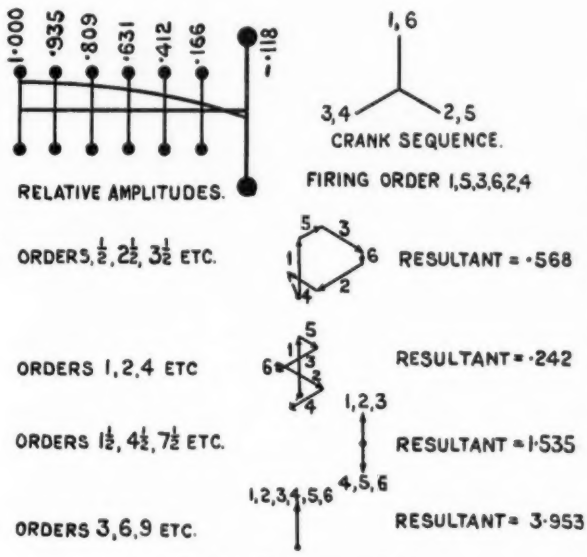


Fig. 3.—Vector summations.

research has shown that the amount of energy thus accounted for is too small to correspond with observed vibration amplitudes. The remaining energy is absorbed in cyclic deformation of the bearing oil films, distortion of the engine structure and in other ways.

A great deal of investigation has been made of observed vibration amplitudes of various engine systems and numerous empirical methods developed for estimating the amount of vibration to be expected in any particular circumstances.

As the engine speed is raised or lowered away from a critical speed the amplitude of vibration decreases so that the variation in induced stress over a range of speeds will take the form illustrated, for a typical case, in Fig. 4. It will be seen that most of the critical speeds in this particular case do not cause stresses of any significant magnitude, so that it would be quite safe to run the engine at any speed below approximately 660 R.P.M. Furthermore, the stress does not build up to the maximum value immediately, so that it is quite safe to pass rapidly through a critical speed even though a fairly high stress would be reached if the engine were to run for any length of time at the resonant speed.

Methods of Reduction of Vibration.

There are various ways of dealing with systems which would otherwise be subject to excessive vibration.

(1) The natural frequency can be altered by modifying the design of some of the parts. This method is clearly not readily applicable to an engine of a range already in production.

(2) The natural frequency can easily be reduced by fitting a small auxiliary flywheel at the end of the crankshaft opposite to the normal flywheel. This method is used on several of the Company's standard engines.

(3) In certain cases, an alteration in firing order will reduce the stress caused by a particular critical speed.

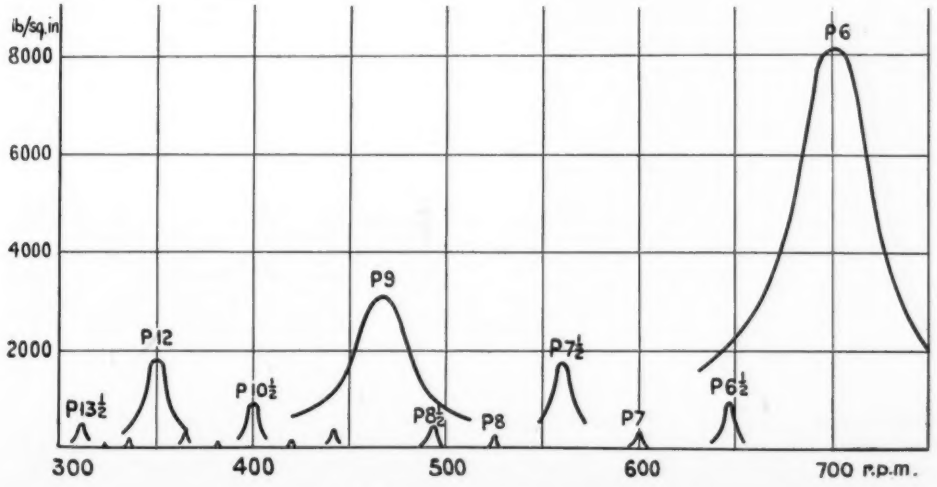


Fig. 4.—Crankshaft stresses caused by torsional vibration.

Fig. 5.—Bibby flexible coupling.

(4) The introduction of a flexible coupling will naturally alter the characteristics of a system considerably. There are various types of coupling in which the torsional stiffness alters with a change in twist and these are very effective in preventing the building up of too great a vibration. The Bibby coupling, which has this characteristic, is shown in Fig. 5.

(5) A fluid coupling or magnetic slip type coupling will isolate one part of a system from another as regards the transmission of torsional vibration. The Company has developed a magnetic coupling which incorporates various special features. An article which describes and illustrates this coupling will be found commencing on page 214 of this issue.

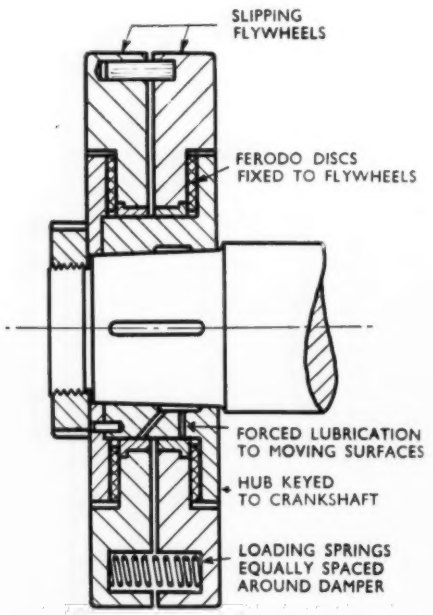

SLIPPING FLYWHEELS
FERODO DISCS FIXED TO FLYWHEELS
FORCED LUBRICATION TO MOVING SURFACES
HUB KEYED TO CRANKSHAFT
LOADING SPRINGS EQUALLY SPACED AROUND DAMPER

Fig. 5a.—" English Electric " H-type engine vibration damper.

Fig. 5b.—" Metalastik " vibration damper.

(6) There are numerous types of vibration damper. The design used on the H-type engine, illustrated in Fig. 5a, consists of two flywheels driven from the crankshaft by friction. If the rotation of the shaft is fairly uniform, the friction torque is sufficient to make the damper flywheels follow the small speed fluctuations of the shaft without any slip. When, however, a critical speed of appreciable severity is reached, the friction torque becomes inadequate and slipping occurs with a consequent absorption of energy. In addition, the fact that the flywheels are no longer rigidly connected to the shaft alters the characteristics of the system and thus changes its natural frequency.

There are other types in which similar results are obtained by forcing oil through orifices when vibration occurs. Another variety, which has been introduced in recent years, consists of a mass flexibly supported by bonded rubber so that it can rotate to some extent in relation to the crankshaft ; a typical design is shown in Fig. 5b.

(7) In aero-engines, a " pendulum " vibration absorber is sometimes used ; this consists of one or more masses freely connected to the

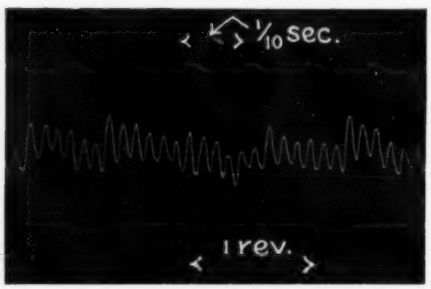


Fig. 6.—Torsiograph record.

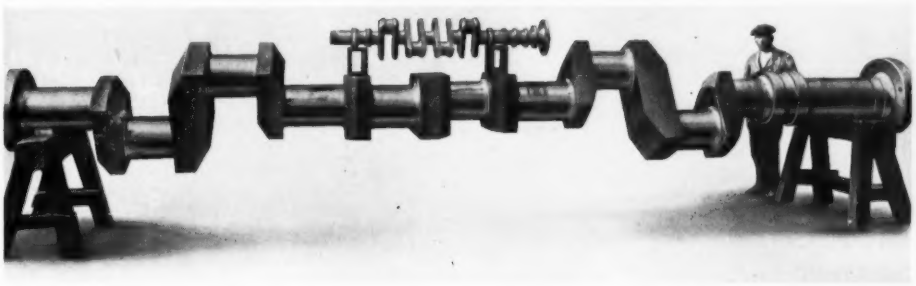


Fig. 7.—Crankshaft from an "English Electric" H-type high-speed Diesel engine, compared with the crankshaft from an R-type Fullagar engine.

crankshaft, their natural vibration frequency being adjusted to a suitable value. It may be of interest to note that the restoring force is caused in this case by centrifugal force, so that the natural frequency is proportional to the rotational speed.

Experimental Technique.

The growing importance of the subject has led to the development of various instruments for recording torsional oscillations. Most of these operate on the usual seismographic principle, the recording being made by an ink trace on paper, by a mark on a celluloid strip or by electrical means using a cathode-ray oscillograph. Some of these instruments also record a time scale and an indication of successive engine revolutions, so that it is possible to determine the engine speed and vibration frequency at any instant. A typical record is shown in Fig. 6. It is also possible to measure the amount of twist at a given point in a shaft by the use of electrical strain gauges and to measure the relative twist between two points on a shaft by a suitable type of torsionmeter.

"English Electric" Oil Engines.

The range of crankshafts dealt with at the Company's Rugby Works is illustrated by Fig. 7. The small shaft is from the first high-speed engine of the H-type and has journals $4\frac{1}{4}$ in. in diameter; the large shaft is from the R-type Fullagar engine and has journals 13 in. in diameter. It will be appreciated that with shafts similar to the latter it is entirely impracticable to determine the torsional stiffness by any experimental method and it is necessary to rely entirely on calculations made in the design stage. The results achieved have been eminently satisfactory.

A Tribute from Tasmania.

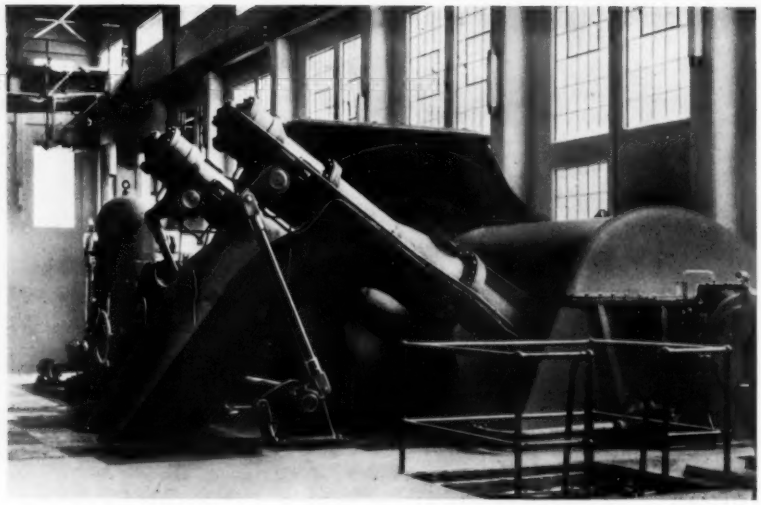
A graceful tribute from Tasmania to the workers of Britain has been received in connection with the recent installation by The English Electric Company of some large hydro-electric generating plant.

This tribute appears in the report issued by The Hydro-Electric Commission of Tasmania for 1943-44, an official Government document, under the heading of "Power Production," and reads as follows :—

> "It is with pleasure that we report a much improved power position. The anxiety that has been felt at heavy peak periods during the past few years has been relieved by the placing into commission of new power units at Tarraleah and Waddamana Stations.
>
> "That Great Britain was able to build for us and to deliver to Tasmania these large turbo-generators during the war whilst fighting for her existence through air and sea warfare is an achievement which we propose to record by a suitable plate on each machine, such as :
>
> 'Built by the workers of Great Britain and carried by British seamen to Tasmania during the war years 1939-1944.'
>
> "Thus these historic units will remain as lasting symbols of the indomitable courage and solidarity of the Empire at a time of great national stress."

The new power units referred to comprise four "English Electric" water turbine driven alternator sets of the horizontal-impulse type, two sets being installed at Tarraleah and two sets at Waddamana.

At the former site each turbine has a rating of 21,000 B.H.P. at a speed of 428 R.P.M., operating with a 940 ft. head. Each alternator has an output of 18,750 kVA. at 11 kV., 3-phase, 50 cycles.



At the latter station the rating of each turbine is 17,600 B.H.P. at 500 R.P.M., with a head of 1,070 ft., and the output of each alternator is 15,000 kVA. at 11 kV., 3-phase, 50 cycles.

The current from these power stations is used for the production of zinc and carbide and an ancillary feature of interest is the inclusion of "English Electric" patented "Aerofoil" water flow recorders in both installations.

One of the 21,000 H.P. water turbine driven alternator sets installed at the Tarraleah Power Station of the Tasmanian Hydro-electric Commission.

Notes on the Design and Operation of Superchargers.

By A. J. PENN, A.M.I.Mech.E., Supercharger and Performance Engineer, D. Napier & Son, Ltd.

(*Abstract of a Paper read before the Royal Aeronautical Society.*)

Supercharging of aero-engines was employed in its early stages to restore the pressure in the induction manifold at altitude to that corresponding to sea level. During recent years, however, due to the improvement of materials and the introduction of higher grade fuels, it has become possible to operate engines at much higher B.M.E.P.'s. As a result of this, supercharging is now employed to boost engines at altitude to pressures well above the sea-level pressure, thus enabling high power outputs at high power to weight ratios to be obtained.

The type of supercharger which has been universally adopted for the modern aero-engine is that of the centrifugal type. This type is essentially a high-speed machine, and therefore for a specific capacity has a low weight ratio and is compact and easy for installation.

The Centrifugal Type of Supercharger.

Briefly the centrifugal supercharger comprises a single- or double-sided rotating impeller enclosed between two half-casings containing fixed diffuser vanes. The impeller is usually gear-driven through increasing gears from the engine, and according to the altitude performance requirements may have a single-speed, two-speed or variable-speed drive.

The alternate method of drive which has recently been perfected after years of research and development is the exhaust-turbine, and this is attractive for attainment of maximum engine power at great heights by utilisation of the energy left in the exhaust gas.

Fig. 1 shows an arrangement of a single-stage centrifugal supercharger. Air inhaled through the intake, "A," enters the impeller, "B," near its axis, and while flowing out through the impeller is given a high tangential velocity. The air then enters the diffuser, "C," in a spiral path between the fixed vanes, "D," during which the kinetic energy imparted to the air is converted into pressure with a rapid reduction of its peripheral velocity. The air leaving the diffuser passes through the delivery volute, "E," to the engine induction system.

Design of the Centrifugal Supercharger.

Inlet Design.—The most suitable design for the inlet to the supercharger is definitely debatable,

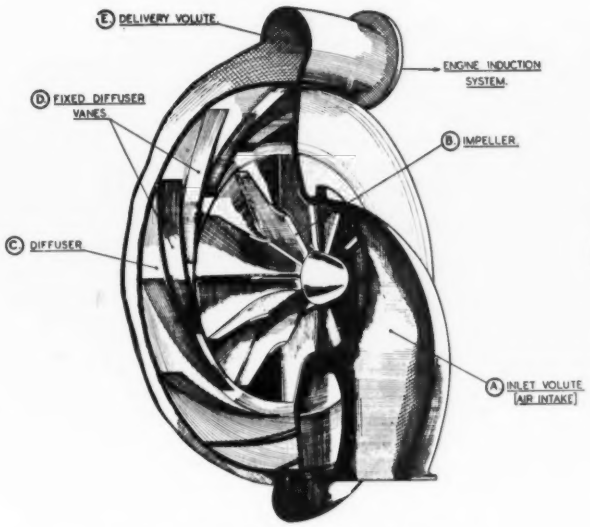


Fig. 1.—Typical arrangement of a single-stage centrifugal triple charger.

but in general it is agreed that the air-flow at the inlet should be axial, whatever design be employed, and evenly distributed on the impeller. Best results are generally obtained with a straight axial intake, but the difference, however, is not great if an intake manifold properly shaped and of ample dimensions be employed.

Impeller Design.—Various designs of impellers are employed in superchargers of different types of aero-engines manufactured in this and other countries ; these types may be classified as follows: double-shrouded ; single-shrouded ; double-entry ; single-entry. These may have integrally bent blades at entry or separate rotating steel guide vanes which may be of the machined or bent type. The single-shrouded, single-entry type impeller with straight radial blades, as shown in Fig. 1, is more generally used for simplicity of manufacture.

With this type of impeller it is essential that the clearance on the bladed side between the edges of the blades and the casing be a minimum, in order to reduce the losses due to leakage from the front side of the blades where the high-pressure region exists, to the low-pressure region at the back of the blades. It is for this reason that the fully-shrouded impeller has a slightly improved maximum efficiency.

It is usual to give some degree of curvature to the blades at entry, in order that the air-flow may enter the impeller practically free from shock. When commencing to design an impeller, it is first necessary to calculate the tip speed for the compression ratio required. The outside diameter, D_2, of the impeller is then decided, in relation to the permissible overall dimensions of the projected supercharger and the maximum safe rotational speed for the degree of required reliability of the supercharger drive.

The number of blades selected for the impeller is a compromise ; a large number reduces the relative eddy velocity and separation between the blades, but results in restriction at the entry of the impeller with resultant high velocity and increased friction. The balance for a specific design can be decided only as a result of experiments. In practice, the number of blades varies between 12 and 20.

Fig. 2 shows the velocity diagrams for the entry guide vanes, assuming a uniform axial flow at the entry, which in practice is usually not quite achieved due to a slight induced swirl.

From the peripheral velocity at entry and the mean axial velocity of the air, it is possible to ascertain the absolute velocity and the entry angle for the entry outer diameter, D_1. From a similar process it is possible to calculate the entry angle at the impeller boss, that is, at the entry inner diameter, d. The angle to which the entry blades should be set at the outer diameter, D_1, is generally 20 to 40 degrees, and that at the inner diameter, d, in the vicinity of the boss,

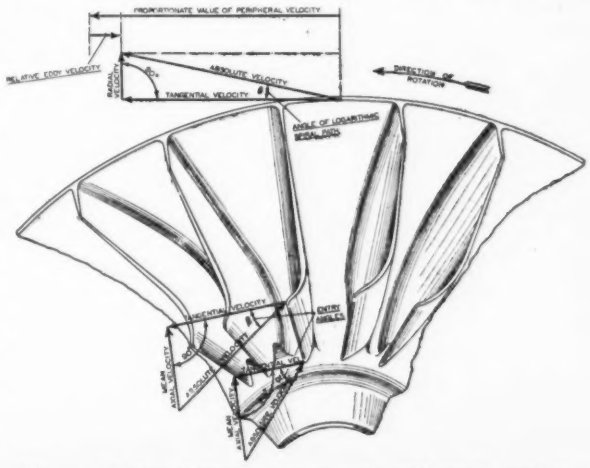


Fig. 2.—Velocity diagrams at entry and periphery of impeller.

30 to 60 degrees. These angles are usually calculated to ensure maximum efficiency for maximum flight performance.

The impeller disc profile is designed to give a safe stress in the disc and the blade contour so that the sectional areas through the channels follow a smooth curve from entry to the periphery.

The area at the periphery of the impeller should be as small as possible, provided that it does not result in too large a log spiral angle. Log spiral angles usually employed range between 10 and 20 degrees, depending upon the altitude at which maximum performance of the supercharger is required.

Fig. 2 shows also the velocity diagram and log spiral angle at the periphery of the impeller, allowing for the effect of the relative eddy velocity but not for separation.

The tangential component of the air leaving the impeller for the radial blade type is less than the peripheral velocity, due to relative eddy. From the tangential velocity and the mean radial velocity it is possible to determine the absolute velocity and the log spiral angle of the air leaving the impeller.

The entry diameter, D_1, of the impeller is decided in relation to the weight of air to be inhaled to give the required engine performance. This is usually between .45 D_2 and .7 D_2, and is to a certain degree affected by the boss diameter, d, for the impeller spindle.

The stressing of the impeller is involved and certain assumptions have to be made. On these assumptions the maximum stress in an impeller is usually at a point on the disc of approximately .6 D_2, and is in the neighbourhood of 6½ tons per square inch for an impeller manufactured in duralumin. Impellers tested to these criteria have proved to be satisfactory.

Fig. 3 shows the typical stress curves for an impeller. Curve A shows the radial stress in the disc and curve B the tangential stress. Regarding the manufacture of impellers, until recently these were machined from a solid duralumin "cake," but they are now machined from die stampings.

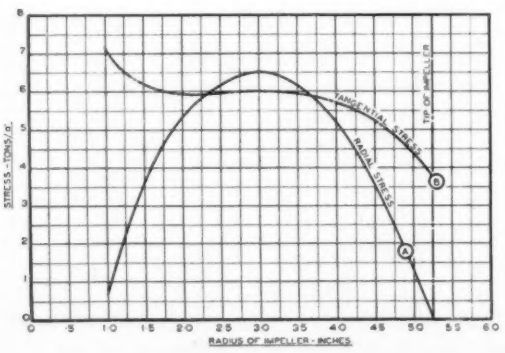


Fig. 3.—Typical stress curves for impeller.

Diffuser Design.—The function of a diffuser with fixed vanes surrounding the impeller at the periphery is to transform the kinetic energy imparted by the impeller to the air into pressure energy as quickly as possible and in the smallest space for compactness.

Fig. 4 shows the air-flow leaving the impeller and entering the diffuser. The energy in the air leaving the periphery of the impeller is partly kinetic and partly pressure.

If the air leaving the periphery of the impeller were allowed, it would form a free vortex co-axial with the impeller, and would flow in a spiral path to the delivery volute exit at the periphery of the casing. The velocity would be reduced and the pressure increased by the laws of vortex motion. To transform efficiently the kinetic energy of a high performance supercharger into pressure by this method would mean reducing the absolute velocity of the air leaving the impeller from 1,000 to 1,200 ft./second to approximately 200 ft./second. This would necessitate a casing of very large dimensions surrounding the impeller, and would be prohibitive in the case of aero-engine design, where compactness and weight are important considerations. It is for this reason that it is general practice in supercharger design for aero-engines to employ a fixed vane diffuser to hasten the transformation of the kinetic energy into pressure.

When a vaned diffuser is employed, it is essential that a free vortex space with converging boundary walls be provided between the impeller and diffuser to steady the air-flow into a uniform condition before entering the passages between the diffuser vanes and to reduce the large pressure fluctuations.

The design of a diffuser can be correct for one condition only, as a change of rate of air-flow at the same impeller tip speed would alter the log spiral angle of the air leaving the impeller, and this would necessitate adjustment of the diffuser angle. It is therefore usual to design a diffuser for the maximum rate of air-flow at the maximum power altitude in flight.

Before commencing to design the diffuser it is necessary to proportion (1) the gap between the impeller and diffuser; (2) the radial depth of the diffuser, and (3) the size of the delivery volute, in relation to the overall dimensions of the supercharger permissible by the layout of the projected engine.

The radial distance between the impeller and diffuser is usually about 0.05 D_2, therefore, D_3, at the entry of the diffuser vanes, is approximately 1.10 D_2, the outer diameter, D_4, of the diffuser vanes usually varies between 1.40 D_2 and 1.60 D_2. The ranges of these ratios in terms of impeller diameter, D_2, are established as a result of previous experiments. The selection of the number

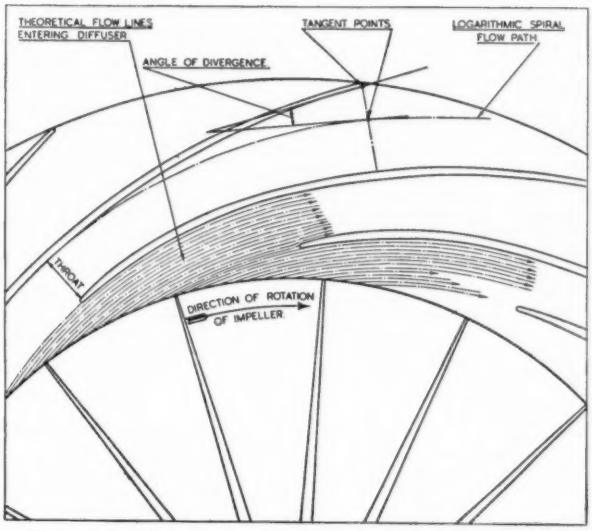


Fig. 4.—Air flow at diffuser entry and divergence between vanes.

of blades for the diffuser is influenced by the delivery volute design which in turn is influenced by the engine induction system, which depends upon the type of engine and the cylinder arrangements.

In practice, the number of vanes varies between eight and twenty, depending upon their thickness and shape and the log spiral angle.

The contour of the vanes is designed to give the required expansion characteristic, previously established as a result of considerable research. The rate of expansion through the diffuser passages is one of the most important factors of the design. The internal contour of the vanes is designed to follow the log spiral angle curve up to the throat formed by the two consecutive vanes, in order to avoid shock to the air at entry. From this point, the vane contours are designed with a certain angle of divergence from the log spiral curve forcefully to divert the air from its free log spiral path to hasten the transformation of the kinetic energy into pressure.

The design of the vanes at the exit should be such that the velocity of the air leaving the diffuser is reduced to within 200 to 300 ft./second.

Delivery Volute Design.—Various designs of delivery volutes are employed for different types of engines and in practically all cases a diffuser with fixed vanes is used between the impeller and volute.

The cross-sections of these delivery volutes may be either rectangular or circular. The circular cross-section, however, is considered to be the most efficient. Careful consideration is necessary in the design of the delivery volute, as generally the conversion of energy at the exit of the vaned diffuser is not complete, and a further recovery of pressure is possible with careful design, as the air passes through the volute. Delivery volute losses, however, although important, are not so serious as intake losses.

Supercharger Performance Calibration.

The determination of the pressure-volume and efficiency characteristics of a centrifugal supercharger on the test-rig necessitates conducting a series of constant speed characteristic curves at different speeds covering the operating range of the supercharger when installed on the engine.

Fig. 5 shows a typical supercharger air-flow calibration, which comprises a series of constant speed characteristic curves covering a range of impeller speeds up to 26,000 R.P.M., with adiabatic efficiency curves superimposed upon the pressure ratio characteristics. Each constant

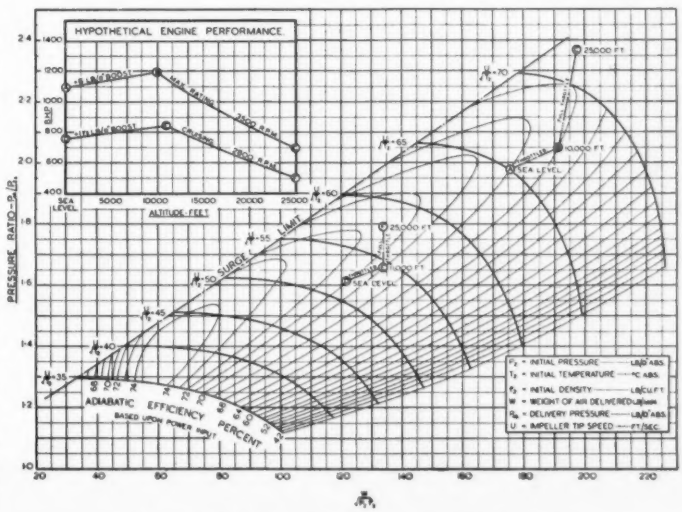


Fig. 5.—Typical supercharger characteristic curves.

speed characteristic curve was commenced at the maximum mass flow, which was reduced in increments by gradually closing the inlet valve until ultimately the minimum flow at surge point was reached. Surging is usually denoted by an abrupt decrease of delivery pressure, and is generally accompanied by audible air vibration.

Surging is an undesirable feature of the centrifugal supercharger and occurs at small rates of air-flow, and is generally accompanied by a sudden breakdown in the relation between the delivery pressure and mass flow.

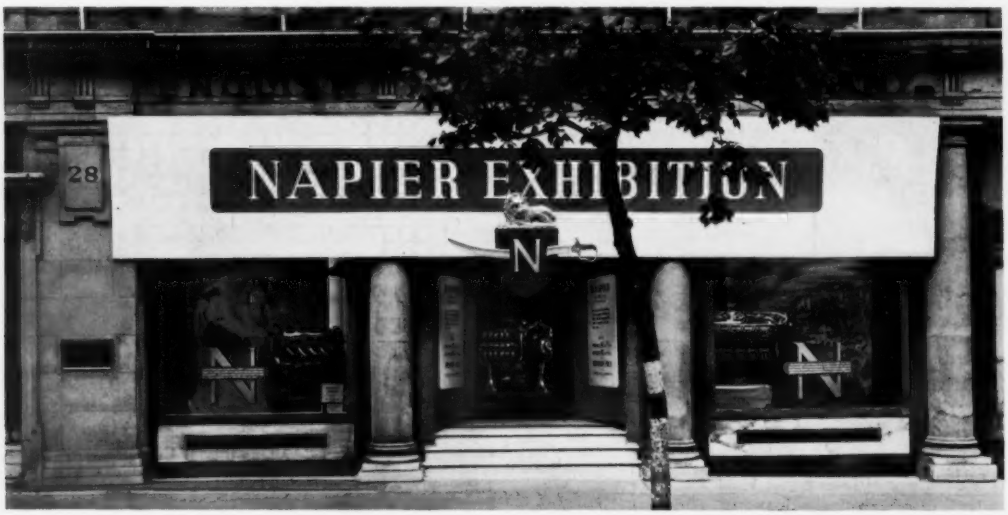
The diffuser entry angle influences the surge point, the smaller angles being more beneficial in postponement of surging.

It will be seen from Fig. 5 that the optimum efficiencies are obtained at the smaller rates of air-flow in the vicinity of the surge point, and therefore, to obtain maximum engine performance, the supercharger should be of such capacity that when installed on the engine the operating points be in this region, but not too close to the surge point, otherwise instability will be experienced.

The diagram at the top left-hand corner of Fig. 5 shows the performance of a hypothetical engine with a maximum rating of +6 lb./sq. in. boost pressure at 3,500 engine R.P.M., and a cruising rating of $+1\frac{1}{2}$ lb./sq. in. boost pressure at 2,800 engine R.P.M. Line AB represents the powers available under throttled conditions of +6 lb./sq. in. boost pressure up to the full throttle height of 10,000 ft., and line BC represents the powers available at full throttle at falling boost pressures up to an altitude of 25,000 ft. Similarly, lines DE and EF represent the powers available for cruising under throttled and full throttle conditions up to an altitude of 25,000 ft. These lines are plotted on the supercharger characteristic curves and indicate the points at which the supercharger would operate if geared to an engine of this capacity and operating under these conditions.

It is apparent from Fig. 5 that when the engine is operating under throttled conditions of constant boost pressure at altitudes above sea level, the supercharger will be operating at increased volumes of air, that is, at higher values of $W/\sqrt{P_2\rho_2}$, with only a slight variation in compression ratio. When the engine is operating at full throttle at altitudes above the full throttle height, the supercharger will be operating at approximately constant volume, that is, at a constant value of $W/\sqrt{P_2\rho_2}$, but with increased compression ratio.

It is important to remember when designing a supercharger that the engine controls the operating point of the supercharger and that the capacities of supercharger and engine must be carefully matched if maximum overall efficiency is to be obtained.



Napier Exhibition in the Showrooms of The English Electric Company, Queen's House, Kingsway, London, which was held from Monday, 28th May, to Friday, 8th June, 1945. Over 20,000 visitors attended the Exhibition.